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Measurement of thermal accommodation coefficients of steel surfaces

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MEASUREMENT OF THERMAL ACCOMMODATION COEFFICIENTS OF STEEL SURFACES

BY

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A

THESIS

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ABSTRACT

The thermal accommodation coefficient plays an important role in low density thermal energy transfer measurement. The object of this investigation was to measure the thermal energy transfer between a heated test surface and a water cooled reference surface (flat black lacquer) consisting of two infinite concentric cylinders separated by dry air.

Two machined and sanded steel cylinders with mean surface roughnesses of 25 microinches and 7.5 microinches were used as the test surfaces. Measurements were made in the pressure range of 1.2×10^{-6} mm Hg. to 1.8×10^{-6} mm Hg. and temperature range for test cylinders of 110^{o} - 200.2^{o}F. in determining the emittance. The pressure range was 1.0×10^{-3} mm Hg. to 1.35×10^{-3} mm Hg. and the temperature range 115.5^{o} - 197.6^{o}F. in determining the thermal accommodation coefficients.

The thermal accommodation coefficient for dry air on a steel surface with an average mean surface roughness of 25 microinches was 0.835 (emittance was 0.174) while for the 7.5 microinches surface condition, the thermal accommodation coefficient was 0.693 (emittance was 0.123).

The experimental data indicated that for the same material, the rougher surface will have a higher value of thermal accommodation coefficient and emittance. The experimental results agree closely with those of classical theory (roughness causes more than one

collision at the surface) and with some other investigators (2 & 7).

The accuracy of the results as well as the experimental deviations are within the accepted engineering limits for this type of measurement.

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NOMENCLATURE

A	Surface area, FT^2
A_o	Avogadro's number, 2.73×10^{26}/LB-MOL
a	Absorptivity
D	Diameter of gas molecule, FT
e	Emissive power, BTU/HR/FT^2
F	Fraction of energy or viewing factor
i, j	Number of integers
K	Thermal Conductivity, BTU/HR/FT/^oF
k	Specific heat ratio
L	Length of inner cylinder, FT
M	Molecular weight of gas molecules
M_o	Molecular weight of air
n	Number of molecules per unit volume
p	Gas pressure, mm Hg.
P	Gas pressure, LB/FT^2
Q_T	Total heat transfer, BTU/HR
Q_R	Radiative heat transfer, BTU/HR
Q_C	Conductive heat transfer, BTU/HR
r_1	Radius of inner cylinder, FT
r_2	Inner radius of outer cylinder, FT
R	Radiative heat transfer per unit area, BTU/HR/FT^2
R_o	Universal Gas Constant, 1545.3 FT-LB/LB-MOL-^oR
^oF	Degree Farenheit (temperature unit)

^{o}K	Degree Kelvin (temperature unit)
^{o}R	Degree Rankine (temperature unit)
T_1	Temperature of inner cylinder, ^{o}R
T_2	Temperature of outer cylinder, ^{o}R
T_g	Temperature of gas, ^{o}R

Symbols

ρ	Reflectivity
ϵ	Emissivity or Emittance
α	Thermal Accommodation Coefficient
λ	Molecular Mean Free Path
σ	Boltzmann constant, 0.1714×10^{-8} BTU-^{o}R/HR-FT^2
σ_c	Effective cross-sectional area of moving molecules, FT^2
π	Constant, 3.14159

I. INTRODUCTION

Thermal conduction through gases can be distinguished by the continuum, transition and rarefied regimes. These regimes are determined by the value of the Knudsen number which is defined as the ratio of the mean free path of the gas molecules to the characteristic dimension of the body or surface.

In the continuum regime ($K_n \leq 0.001$), where the molecular mean free path is several orders of magnitude smaller than the characteristic dimension, thermal conduction through a gas is the result of numerous collisions between gaseous molecules. Hence, in classical continuum theory, thermal energy transfer is the transfer of kinetic energy from one molecule to another by inter-molecular collisions rather than individual molecular collisions with the solid surfaces.

In the rarefied regime ($K_n \geq 10$), where the molecular mean free path is large compared to the characteristic dimension, inter-molecular collisions between gas molecules are infrequent when compared with molecular collisions with solid surfaces.

In the transition regime ($0.001 \leq K_n \geq 10$), the molecular mean free path of the gas is of the same order as the characteristic dimension. This regime is in between continuum and rarefied regimes. Thermal energy is transferred by both the inter-molecular collisions and the collisions on the solid surfaces. Knowledge of the thermal accommodation coefficient at the solid surface is necessary for the calculation of thermal conduction in this regime. However, experimental measurement of the thermal accommodation coefficient is usually required for complete evaluation of energy transfer in this regime.

For a low density gas (transition or rarefied regimes), thermal energy transfer between solids takes place not only by conduction but also by radiation. Thermal radiation is the ability of a body or surface to emit energy in the form of electromagnetic waves. The amount of thermal energy emitted by a surface as radiation is dependent upon the value of a surface property called emissivity, ϵ , as well as other characteristics.

The emissivity (or emittance) is the ratio of the total emissive power of the body or surface to the total emissive power of a black body at the same temperature. Accurate values of emittance for any body or surface require experimental determination due to the strong influence of surface characteristics.

Thermal conduction through low density gases is dependent upon the gas pressure, thermal accommodation coefficient, molecular weight, gas temperature, and the difference in temperature between the solid boundaries. The thermal accommodation coefficient, α , is defined by equation 1,

$$\alpha = \frac{E_1 - E_2}{E_1 - E_s} \qquad \text{---------------------} \; (1)$$

where E_1 is the incident thermal energy of the gaseous molecule, E_2 is the thermal energy of the reflected or re-emitted molecule and E_s

is the thermal energy of the surface molecule. If the gaseous molecule comes to complete thermal equilibrium with the surface molecule, the molecule is said to be "completely accommodated" such that the thermal accommodation coefficient is equal to one. For typical gases and surfaces $0 < \alpha < 1$.

Difficulty arises in determining an accurate value for thermal accommodation coefficient since it depends on many parameters. Although by definition it is independent of gas pressure, the value of the thermal accommodation coefficient appears to change as surface characteristics changes with pressure due to adsorbed gases at the surface. In particular, the thermal accommodation coefficient not only depends on the structure of the gas molecule itself, but also the physical and mechanical condition of the surface.

Many investigators have reported quite different values for the same solid surface-gas combinations (1)[1]. One of the reasons for these deviations was that the surface conditions were not closely controlled nor evaluated in the experiments. In conducting the experiment, "ageing" affects the value which is measured for the thermal accommodation coefficient. This ageing effect is due to the decomposition of materials on the surface and the adsorption and emission of gases on the surface during the execution of the experiment. Probably errors have been indicated by some investigators for the measured values of the thermal accommodation coefficient. Obviously, the value of the

[1] Numbers in ( ) refer to references given in Bibliography.

thermal accommodation coefficient must be used carefully in the calculation of the heat transfer by conduction in low density gases.

In this experimental investigation, it was intended to show the effect of the mechanical surface condition on the value of the thermal accommodation coefficient. It was also the purpose of this investigation to extend the work done by Dethorne (2) and to improve his experimental apparatus by making some simple modifications. A steel cylinder with various mechanical surface conditions was used to measure the differences in the value of the thermal accommodation coefficient. It was intended to have the steel surface as close to typical engineering (rough and unclean surface) surfaces as possible.

In recent years, only limited experimental data of questionable accuracy for the thermal accommodation coefficient were available for determining the thermal conduction in rarefied gases. For wind tunnel and space simulation chamber testing, values of the thermal accommodation coefficient affect the accuracy of pressure measurements at orifices and along the connecting tubes. The investigation of thermomolecular pressures at orifices by Kinslow and Arney (3) indicated that an accurate value of the thermal accommodation coefficient is necessary to determine the true value of pressure. The value of the thermal accommodation coefficient is also essential for a reasonable estimate of thermal energy transfer for re-entry vehicles during certain regimes of space flight.

II. REVIEW OF LITERATURE

The thermal accommodation coefficient may also be expressed by replacing the energies of the molecules with the absolute temperature, if only mean translational energy changes are considered for molecules having a Maxwell-Boltzmann distribution. However, molecules are not completely accommodated to the surface temperature during a single collision (4). Consequently, the expression of Eq. (1) is the more correct and most useful equation for defining the thermal accommodation coefficient. The two basic methods used for measuring the thermal accommodation coefficient are the slip-flow temperature jump method and the low pressure free molecule flow method.

In the slip-flow regime, where the Knudsen number is between 0.001 and 0.1, there is a temperature discontinuity at the gas-solid surface. The magnitude of this discontinuity is used to determine the thermal accommodation coefficient. This method is not well developed and the accuracy of the measurement of the temperature discontinuity needed for determining the thermal accommodation coefficient is far from being satisfactory. As has been pointed out by Devienne (4), for the case of platinum, the ratio of the temperature jump distance to the molecular mean free path is 1.436 for helium, and the calculated thermal accommodation coefficient value is 0.149. However, the range of the thermal accommodation coefficient from his experimental analysis was between 0.146 to 0.196. If the value of 0.196 for the thermal accommodation coefficient is used, the temperature jump distance is about one third smaller. Therefore, this method of measuring the thermal accommodation coefficient has not been used to any great extent and the results from these measurements are considered unreliable.

The low pressure free molecule method for measuring the thermal accommodation coefficient is more reliable. Hence, this method has been adapted and evaluated by many investigators. Also, many theoretical equations for evaluating the thermal accommodation coefficient have been proposed. Yet, they are only applicable in very special cases due to the fact that they neither account for variation of the thermal accommodation coefficient in terms of the angle of incidence of the molecules, nor for the nature and the amount of the gas adsorbed by the surfaces. Mann (5) has done some investigation of the adsorbed gas film on a platinum surface due to impurities in the gas. He found that for helium, the thermal accommodation coefficient was 0.03 at room temperature and 0.04 at $80^{\circ}K$ for a range of mean filament temperature between 100° and $1000^{\circ}C$.

Descriptions of the apparatus and method of measurement using the temperature jump method and the low-pressure free molecule method are given by Dethorne (2), Hartnett (6), and Wachmann (1).

Wiedmann and Trumpler (7) investigated the measurement of the value of the thermal accommodation coefficient of air on metallic surfaces and painted surfaces. The apparatus which was used consisted of two concentric cylinders having different surface properties.

The heated center cylinder was used as a test cylinder whereas the water cooled outer cylinder was the reference cylinder. They found that the emissivity for flat black lacquer was 0.932 and the thermal accommodation coefficient was 0.888. For machined cast iron, the emissivity was 0.391 and thermal accommodation coefficient was between 0.87 to 0.93.

The experimental apparatus used by Dethorne (2) was similar to that used by Wiedmann and Trumpler (7). He found that for flat black lacquer, the emissivity was 0.965 and the thermal accommodation coefficient was 0.960. For machined steel, the emissivity was measured at 0.1325 and the thermal accommodation coefficient was 0.971. There was no apparent relationship between emittance and the thermal accommodation coefficient when air was used as the gas between two concentric cylinders.

Kinslow and Arney (3), in their investigation of thermo-molecular pressure effects, found that the value of the thermal accommodation coefficient is important for determining the true value of pressure. As an example, they considered the case of helium at a high temperature over a 1/8 inch diameter orifice in a plane surface at 300°K connected by a 0.25 inch diameter tube to pressure sensing device at 300°K. They calculated that the device will read a 22% error in pressure if the thermal accommodation coefficient equals to 0.3 whereas it will read an 8% error if the thermal accommodation coefficient equals to 0.9. Accurate values are a necessity therefore for accurate pressure measuring systems in heated low density environments. In their experiment, the measured thermal accommodation coefficient between aluminum and copper surfaces for hydrogen was 0.42, for helium was 0.51, for argon was 0.83 and for nitrogen was 0.79.

The investigation conducted by Teagan and Springer (8) was in the transitional regime where the Knudsen number was between 0.001 and 10. Both heat conduction and density distributions were measured for argon and nitrogen between aluminum surfaces. The value of the thermal accommodation coefficient measured for an aluminum surface with argon was 0.826 and for nitrogen was 0.76.

Apparently, only Wiedmann and Trumpler (7) and Dethorne (2) have taken measurements on materials close to engineering interests. The test surface conditions were inadequately defined since it is difficult to describe or measure surface conditions. However, it affects the measured value of the thermal accommodation coefficient strongly. The comparison of the measured values of the thermal accommodation coefficient by various investigators are shown in Table I. Because of the lack of knowledge of the surface conditions for measured values of the thermal accommodation coefficient, the validity of existing data is questionable.

TABLE I
COMPARISON OF MEASURED VALUES OF
THERMAL ACCOMMODATION COEFFICIENT

Investigator (Ref #)	Material	Gas	Temperature °R	Surface Condition	Thermal Accommodation Coefficient
Devienne (4)	Platinum	Helium	Unspecified	Unspecified	0.146 - 0.149
Mann (5)	Platinum	Helium	Room Temperature	Unspecified	0.03
Wiedman and Trumpler (7)	Bronze	Air	578.7 - 618.1	Painted With Flat Black Lacquer	0.881-0.894
	Cast Iron	Air	575.2 - 592.9	Machined	0.87 - 0.88
	Cast Iron	Air	587.5 - 604.9	Polished	0.87 - 0.93
Dethorne (2)	Steel	Air	558 - 599.3	Painted With Flat Black Lacquer	0.960
	Steel	Air	551 - 584	Machined Approx. 50 Microinches	0.971

Continued on next page

Table I (continued)

Investigator (Ref #)	Material	Gas	Temperature °R	Surface Condition	Thermal Accommodation Coefficient
Kinslow and Arney (3)	Aluminum and Copper	Hydrogen	Unspecified	Unspecified	0.42
		Helium	Unspecified	Unspecified	0.51
		Argon	Unspecified	Unspecified	0.83
		Nitrogen	Unspecified	Unspecified	0.79
Teagan and Springer (8)	Aluminum	Argon	Unspecified	Unspecified	0.826
		Nitrogen	Unspecified	Unspecified	0.76

III. EXPERIMENTAL ANALYSIS

Because of the difficulty in expressing the surface characteristics analytically and thereby relating their effect to the basic parameters of low density thermal energy conduction, an experimental investigation has been conducted.

The object of the experimental investigation was to measure the thermal energy transfer between two concentric cylinders separated by dry air at a low pressure. The heat was supplied by an electric heating coil placed inside the center cylinder whereas the outer cylinder was cooled by the circulation of water through copper tubing soldered to the outside of the outer cylinder.

By maintaining the outer cylinder surface in a constant condition for all heat transfer measurements, this surface can then be used as a reference surface where the thermal accommodation coefficient and emittance are known. The measurement of the thermal accommodation coefficient and emittance for this reference surface can be accomplished through a certain experimental procedure which is described in Section IV.

Then, the various inner cylinder surface conditions, measurements of the thermal accommodation coefficient and emittance can be made. The emissivity, (emittance), must be measured for all surfaces since thermal energy transfer by radiation is a major mode of energy exchange during low density heat transfer. By coating the reference surface with flat black lacquer, it becomes nearly "gray" in response to radiation.

A. DERIVATION OF EQUATIONS

For two concentric cylinders at different temperatures separated by a small space which is filled with a low density gas, the total heat transfer between the surface is given by,

$$Q_T = Q_R + Q_C \qquad \text{------------------(2)}$$

where Q_R is the thermal energy transferred by radiation and Q_C is the thermal energy transferred by molecular conduction of the separating gas. By assuming that the cylinders are infinitely long (concentric cylinder clearance $<<$ cylinder length), Q_R and Q_C can be analytically determined.

For enclosure consisting of "gray" surfaces (9)

$$R_i = e_i + \rho_i \sum_{j=1}^{N} R_j F_{ij} \qquad \text{-------------(3)}$$

where i and j are representing the particular surfaces and,

R = Radiative heat transfer per unit area

e = Emissive power of the surface

ρ = Reflectivity

F = Viewing factor

N = Number of surfaces considered

The radiation heat transfer between infinite "gray" concentric cylinders with non-absorbing media is,

$$\frac{Q_R}{A_1} = R_1 - R_2 \qquad \text{-----------------(4)}$$

where

$$R_1 = e_1 + \rho_1 \sum_{j=1}^{2} R_j F_{1j} \text{ ----------------------------(5)}$$

$$R_2 = e_2 + \rho_2 \sum_{j=1}^{2} R_j F_{2j} \text{ ----------------------------(6)}$$

where subscript 1 refers to the test surface and subscript 2 indicates the reference surface. The symbol A denotes surface area.

To calculate Q_R / A_1 the following relations are used:

$$A_1F_{12} = A_2F_{21} \text{ -------------------------(7)}$$

$$\sum_{j=1}^{2} F_{ij} = 1 \text{ -------------------------(8)}$$

$$e = a\, e_b \text{ --------------------------------(9)}$$

$$\epsilon = a \text{ -----------------------------------(10)}$$

In the above equation where e_b is representing the emissive power of the black body and is equal to σT^4; and,

σ = Boltzmann constant

T = Absolute temperature

a = Absorptivity

ϵ = Emittance or Emissivity

Since the transmissivity is zero,

$$\rho + a = 1 \text{ --------------------------(11)}$$

Substituting Eqs. (5), (6), (7), (8), (9), (10), (11) into Eq. (4) yields,

$$\frac{Q_R}{A_1} = \frac{1}{\frac{1}{\epsilon_1} + F_{21}\left(\frac{1}{\epsilon_2} - 1\right)} \sigma \left(T_1^{\,4} - T_2^{\,4} \right) \text{ ---(12)}$$

and since,

$$A_1 = 2 \pi r_1 L \quad \text{------------------------(13)}$$

$$A_2 = 2 \pi r_2 L \quad \text{------------------------(14)}$$

$$F_{21} = \frac{A_1}{A_2} \quad \text{----------------------------(15)}$$

where

r_1 = Radius of inner cylinder

r_2 = Inner radius of outer cylinder

L = Length of the test cylinder

The radiation heat transfer equation becomes,

$$Q_R = \frac{2 \pi r_1 L \sigma}{\frac{1}{\epsilon_1} + \frac{r_1}{r_2} \left(\frac{1}{\epsilon_2} - 1 \right)} \left(T_1^4 - T_2^4 \right) \quad \text{------------(16)}$$

The equation for heat conduction between concentric cylinders at low density (assuming complete accommodation) was developed by Knudsen. The equation is given as (7):

$$Q_C = 3600 \, r_1 L \frac{k + 1}{k - 1} \frac{p \left(T_1 - T_2 \right)}{\sqrt{M T_g}} \quad \text{-------------------(17)}$$

where

k = Specific heat ratio of gas

M = Molecular weight of gas

p = Pressure of gas

T_g = Absolute temperature of gas

and the units are given in the Nomenclature.

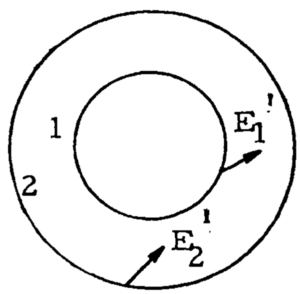


Fig. 1 Low Density Thermal Conduction Model (7)

The equation was extended by Wiedmann and Trumpler (7). They considered a large number of gas molecules striking a unit area of surface 1 (see Fig. 1), with energy E_2' before collisions and E_1' after collisions. The accommodation coefficient at surface 1 is given by,

$$\alpha_1 = \frac{E'_2 - E'_1}{E'_2 - E_1} \quad \text{------------------------------(18)}$$

After the molecules collide with surface 2 (see Fig. 1), molecules with energy E_1' have only a fraction "F" of the energy of the molecules (E_1') approaching the outer surface. The accommodation coefficient at surface 2 is given by,

$$\alpha_2 = \frac{F E'_1 + (1 - F) E'_2 - E'_2}{F E'_1 + (1 - F) E'_2 - E_2} \quad \text{--------------------(19)}$$

However, $(E_1 - E_2)$ is the amount of conduction heat transfer as expressed by Knudsen (Eq. 17) and $(E_1' - E_2')$ is the actual amount of heat transfer by conduction. Therefore, Eq. (17) can be expressed by,

$$Q_C = E_1' - E_2' \qquad \text{------------------------------------ (20)}$$

or

$$Q_C = 3600 \; r_1 \; L \; \frac{k+1}{k-1} \; \frac{p\,(T_1 - T_2)}{\sqrt{MT_g}} \; \frac{1}{F\left(\frac{1}{\alpha_2} - 1\right) + \frac{1}{\alpha_1}} \qquad \text{---(21)}$$

The fraction of energy "F" can be found by the same theory as the viewing factor. Thus,

$$F = \frac{A_1}{A_2} = \frac{r_1}{r_2} \qquad \text{-------------------------------(22)}$$

Substituting Eq. (22) into Eq. (21), yields the thermal energy conduction equation at low density,

$$Q_C = 3600 \; r_1 \; L \; \frac{k+1}{k+1} \; \frac{p\,(T_1 - T_2)}{\sqrt{MT_g}} \; \frac{1}{\frac{r_1}{r_2}\left(\frac{1}{\alpha_2} - 1\right) + \frac{1}{\alpha_1}} \qquad \text{----(23)}$$

When the gas molecules are moving in a Maxwellian velocity distribution, the molecular mean free path is defined as (10 & 11):

$$\lambda = \frac{1}{\sqrt{2}\; n \; \sigma_c} \qquad \text{-----------------------------(24)}$$

where

λ = Molecular mean free path

σ_c = Effective cross-sectional area of moving molecules

n = Number of molecules per unit volume

For a perfect gas, Eq. (24) can be expressed by,

$$\lambda = \frac{R_o \; M \; T_g}{\sqrt{2}\,\pi \; A_o \; D^2 \; M_o \; P} \quad \text{--------------------(25)}$$

where

R_o = Universal gas constant

M_o = Molecular weight of air

A_o = Avogardro's number

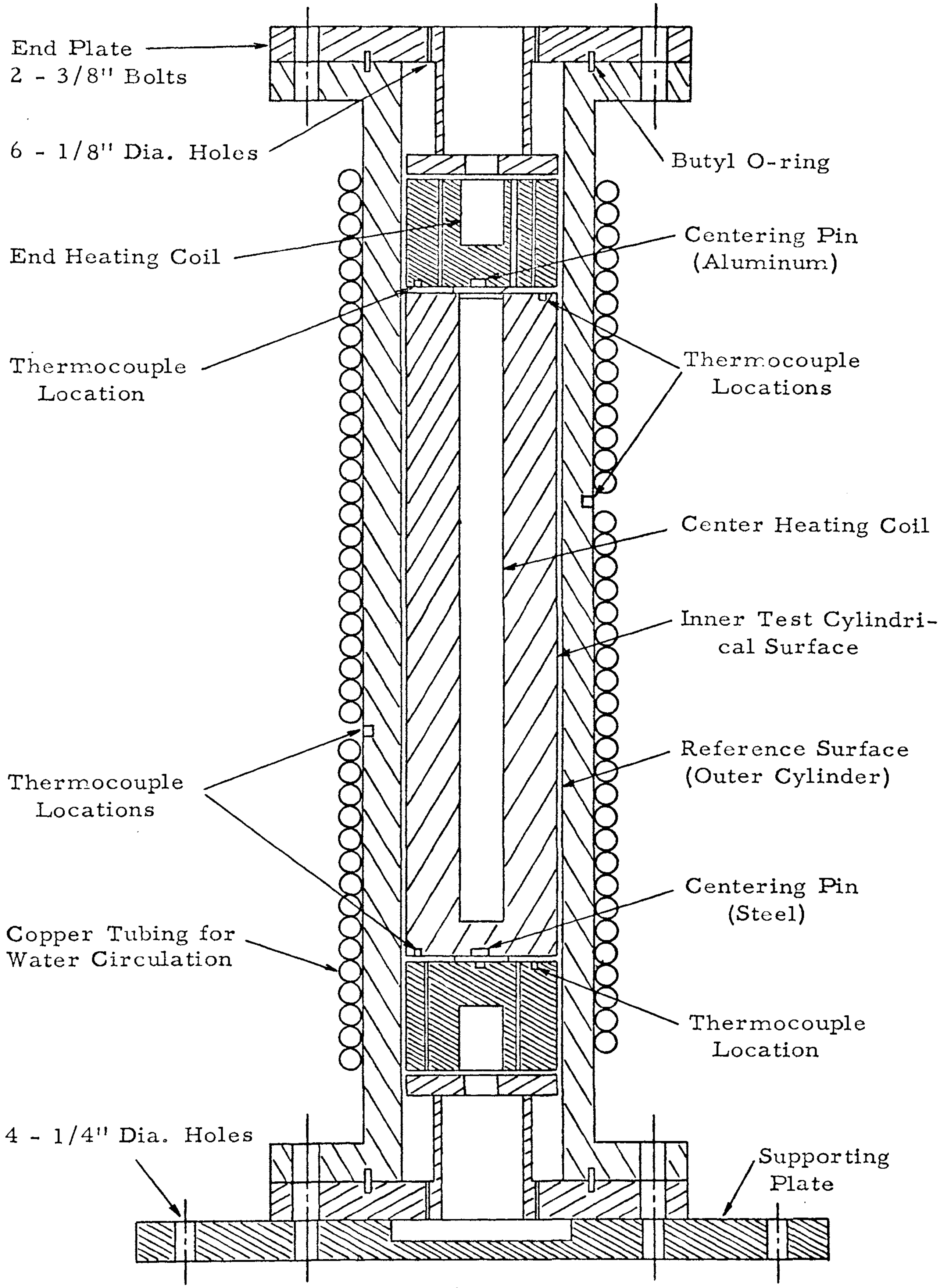
D = Diameter of gas molecules

P = Pressure of gas

B. DESCRIPTION OF APPARATUS

The experimental apparatus used for the measurement of emittance on steel surfaces was similar to that used by Dethorne (2). The apparatus is shown in Fig. 2. The heated center cylinders are 1.91 inches in diameter by 9.0 inches long. The diameter of the concentric outer cylinder is 1.984 inches giving an average separation space of 0.037 inch. The outer cylinder is coated with flat black lacquer and was used as the reference surface. Cooling coils (3/8 inch diameter copper tubing) are soldered around the outside of the outer cylinder for water cooling.

End plug heaters are in place at each end of the center test cylinder so that heat losses from the center cylinder could be minimized. A centering pin is located in each of the end plugs to insure that the test



Half Scale

Fig. 2 Assembly of Test Unit

cylinder does not come in contact with the outer surface. Holes were drilled in each end of the apparatus to allow free movement of the molecules into and out of the separating space.

Six Iron-Constantan thermocouples are located in the apparatus for indicating temperature at critical locations.

C. EQUIPMENT USED

The surface roughness of the test cylinder was measured by a Profilometer with Amplimeter type QB, Model S23, Serial No. 2791, 115 volts, 50-60 cycles and Pilotor type VB, Model 5, Serial No. 1944, 115 volts, 60 cycles made by Micrometrical Division, Ann Arbor, Michigan.

The center heating coil was a Hotwatt Model No. 6948 rated at 60 watts and 400 volts. The power to this heater was supplied by a Heathkit Regulated D. C. Power Supply, Model PS-4, 400 volts at 100 milliamperes (125 milliamperes maximum) made by the Heath Company, Benton Harbor, Michigan. An external shunt made by Weston Electrical Instrument Company rated at 50 amperes and 50 millivolts was installed in series to the power supply to measure the current to the center heating coil. A Hewlett-Packard 419A D. C. Null Voltmeter, Serial No. 532-00489 was also used to measure the power input to the center heating coil.

The end plug heating coils are Hotwatt Model No. 6948 rated at 40 watts, 115 volts each. The power input was supplied by two Powerstat

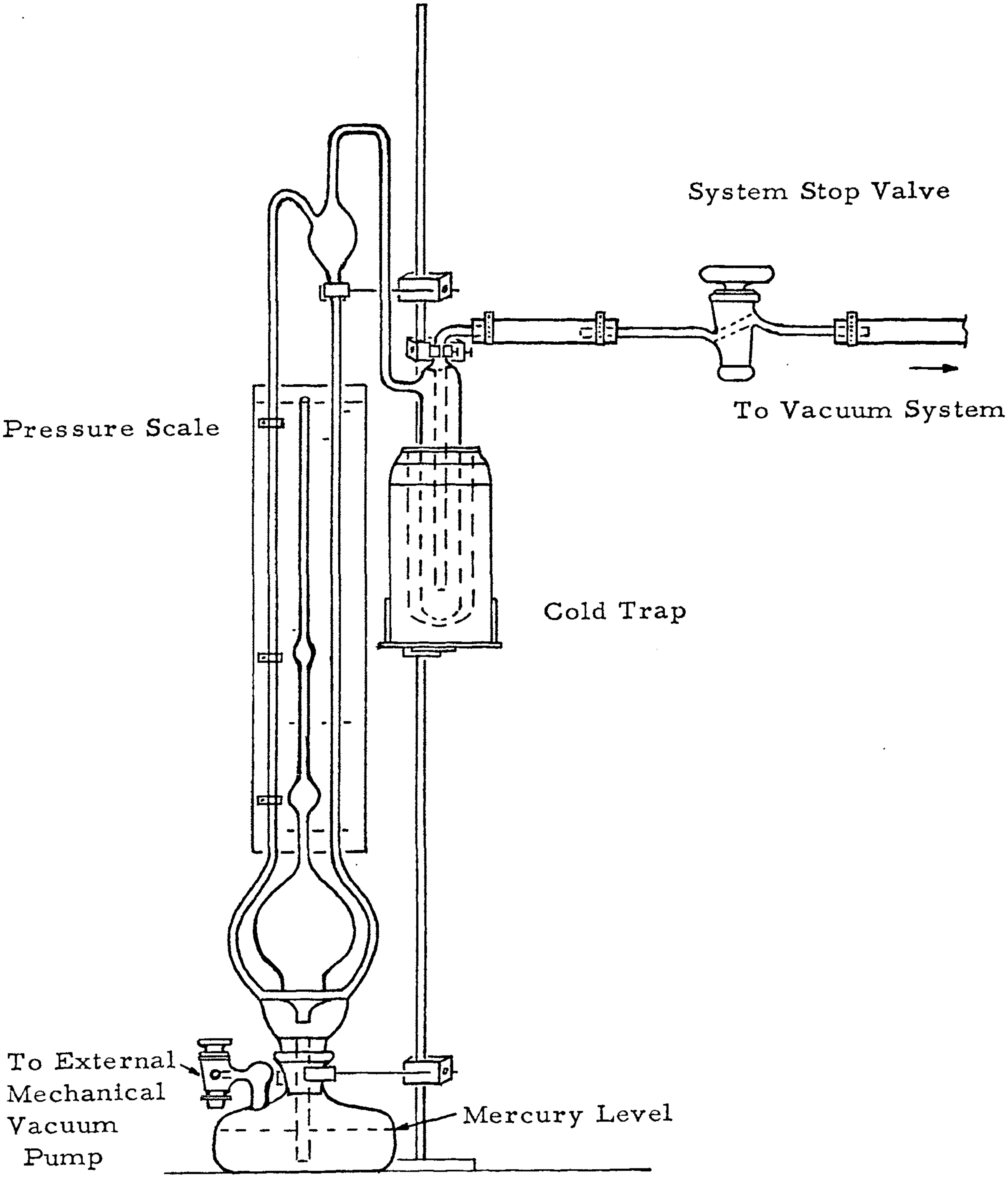
Variable Autotransformers Type 116B, rated from 0-140 volts and 10 amperes made by Superior Electric Company, Bristol, Connecticut.

A Honeywell Potentiometer, Model 2745, Serial No. P-8620 was used for temperature measurement.

The vacuum system was a Varian Vacuum Model VE-61 equipped with a mechanical and a diffusion pump. An ionization gage IG-10 and a thermocouple gage were mounted on the panel of the unit. The vacuum system has the capability of maintaining a vacuum down to 10^{-7} Torr with the use of a liquid nitrogen baffle system.

For more accurate pressure readings, a McLeod Gage Type GM-100A with a range of 0.01 micron Hg. to 10 Torr made by the Consolidated Vacuum Corporation, Rochester, New York, was connected to the vacuum system. A cold trap, an isolation valve, and an external Mechanical Duo-Seal Vacuum Pump, Model 1402, Serial No. 55574 made by The Welch Scientific Company were installed to complete the system. Fig. 3 shows the schematic diagram of the McLeod Gage and its connections.

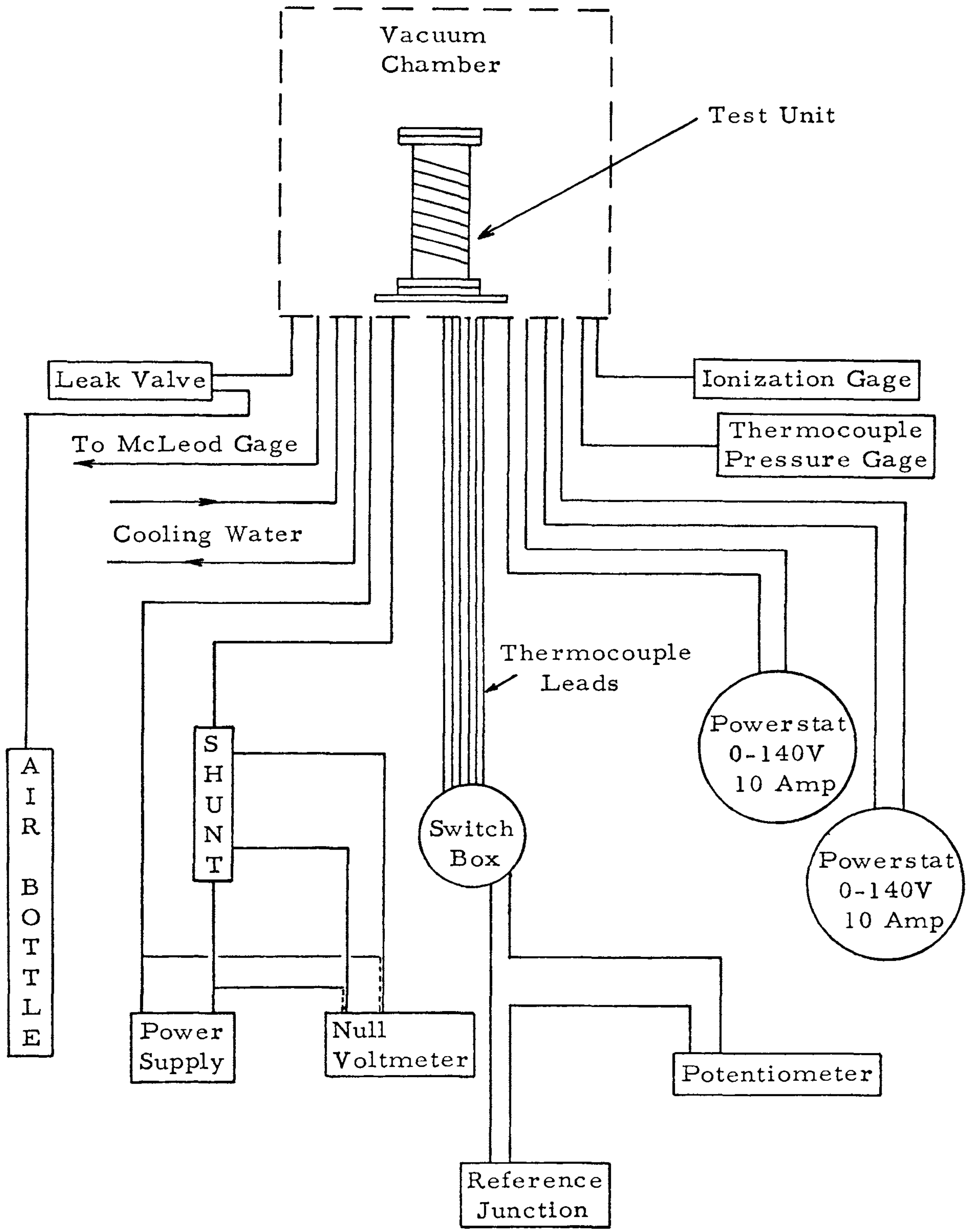
The residue inside of the McLeod Gage was cleaned with a chromic acid solution (solution of sodium dicromate $Na_2Cr_2O_7$ and concentrated sulphuric acid H_2SO_4) followed by rinsing with distilled water. The residue was found to reduce the accuracy of pressure measurement. The above cleaning procedure was therefore used. Although the gage range is to 0.01 micron Hg., the accuracy is doubtful when the pressure is lower than one micron Hg.



Scale: 1" = 5"

Fig. 3 McLeod Gage Installation

A variable leak valve, with a maximum throughput of 100 standard c.c. per second, Series 203 made by Granville-Phillips Company was used to maintain the pressure within the vacuum system at the desired level. Dry air was supplied by an air bottle through the leak valve to the system when needed. The complete wiring diagram is shown in Fig. 4.



No Scale

Fig. 4 Wiring Diagram

IV. EXPERIMENTAL PROCEDURES

In order to measure the emittance and the thermal accommodation coefficient for the reference surface of the outer cylinder, a black steel cylinder with identical surface conditions was initially installed in the apparatus. The apparatus was then placed inside the bell jar chamber of the Varian Vacuum Unit. This configuration gave equal values of emittance and the thermal accommodation coefficient for the two thermal energy exchange surfaces.

A. MEASUREMENT OF EMITTANCE FOR THE REFERENCE SURFACE

A large number of tests were run at the lowest possible pressure inside the vacuum system (approximate 10^{-7} Torr). Under these conditions the gas density was so low that Q_C was approximately zero. Other tests were conducted at slightly higher pressures in order to determine the effect of pressure on Q_C. According to Soddy and Berry (12), the conduction distribution is proportional to the pressure.

At these low pressures, the temperature response of the unit was extremely slow. Power adjustments were made to the center cylinder heater and the end plug heaters to give the desired surface temperature while water flow was maintained for cooling the outer surface. Adjustments were made to the power and temperatures were measured and recorded every hour. A time period of from twelve to eighteen hours was required to obtain satisfactory temperature equilibrium for each test run. By correcting the data for end plug heat flow according to the method given in Appendix C and using equation (16) with $\epsilon_1 = \epsilon_2$, values of emittance were measured. The data for these test runs are given in Table IV in Appendix A.

The temperature of the inner cylinder ranged from 73.1° - 200.2°F. The water temperature, which was assumed to be the same as that of the reference surface ranged from 58.5° - 62.5°F. Pressure for the radiation tests ranged from 5.8×10^{-5} mm Hg. to 1.2×10^{-4} mm Hg.

The mean diameter of the reference surface cylinder was 1.911 inches and had an average surface roughness of approximately 50 micro-inches.

B. MEASUREMENT OF THE THERMAL ACCOMMODATION COEFFICIENT FOR THE REFERENCE SURFACE

Many tests were conducted with the same test unit configuration but at higher pressure, where Q_C was not negligible (one to ten microns Hg.). By similar data correction for end losses of the center cylinder (Appendix C) and by use of equations (2), (16), and (23), ($\epsilon_1 = \epsilon_2$; $\alpha_1 = \alpha_2$; $Q_{measured} = Q_R + Q_C$), values of the thermal accommodation coefficient at different temperatures (70° F. to 200° F.) were measured. Each of these tests required approximately eight to twelve hours to complete. The results from these tests are given in Table V in Appendix A.

C. MEASUREMENT OF EMITTANCE AND THE THERMAL ACCOMMODATION COEFFICIENT FOR STEEL SURFACES

Having the emittance and the thermal accommodation coefficient of the reference surface, the center cylinder was replaced by a machined steel cylinder of the same dimensions (1.910 inches diameter), but first with an average surface roughness of 25 microinches and then with an average surface roughness of 7.5 microinches. These surface conditions were obtained by using 120, 220, and 320 grit emery papers.

Tests were conducted for measuring emittance and then for measuring the thermal accommodation coefficient by using the same technique as given above. The results from these tests are given in Table VI through IX in Appendix A.

V. DISCUSSION

In order to approach the condition of infinite cylinders with no axial heat conduction, the temperature difference between the test cylinder ends and the end plugs was minimized. However, there was still an appreciable amount of heat transfer even for slight temperature differences. For example, a temperature difference of 0.5° F. resulted in an approximate axial heat conduction of 5.32 BTU/HR for the steel centering pin and 24.2 BTU/HR for the aluminum centering pin. Corrections for these losses were made to all of the tabulated results given in Appendix C. These corrections were made according to the values given in Table IV through IX in Appendix A.

A. REFERENCE SURFACE VALUES

The radiation test results from Table IV are plotted in Fig. 5 as a function of cylinder temperature. A least squares fit for the results was attempted but the curve fit showed less than two per cent change in emittance as the temperature was increased so that it was assumed that the average value of $\epsilon = 0.958$ was constant and valid for the entire temperature range investigated.

Several test points have been omitted from Fig. 5 due to their obvious inaccuracies. This average emittance value is quite close to the value obtained by Dethorne (2) for the same cylinder ($\epsilon = 0.965$).

Fig. 6 shows that the results from Table V in Appendix A for the thermal accommodation coefficient at the reference surface. Again, a least squares curve indicated only about a two per cent change in the

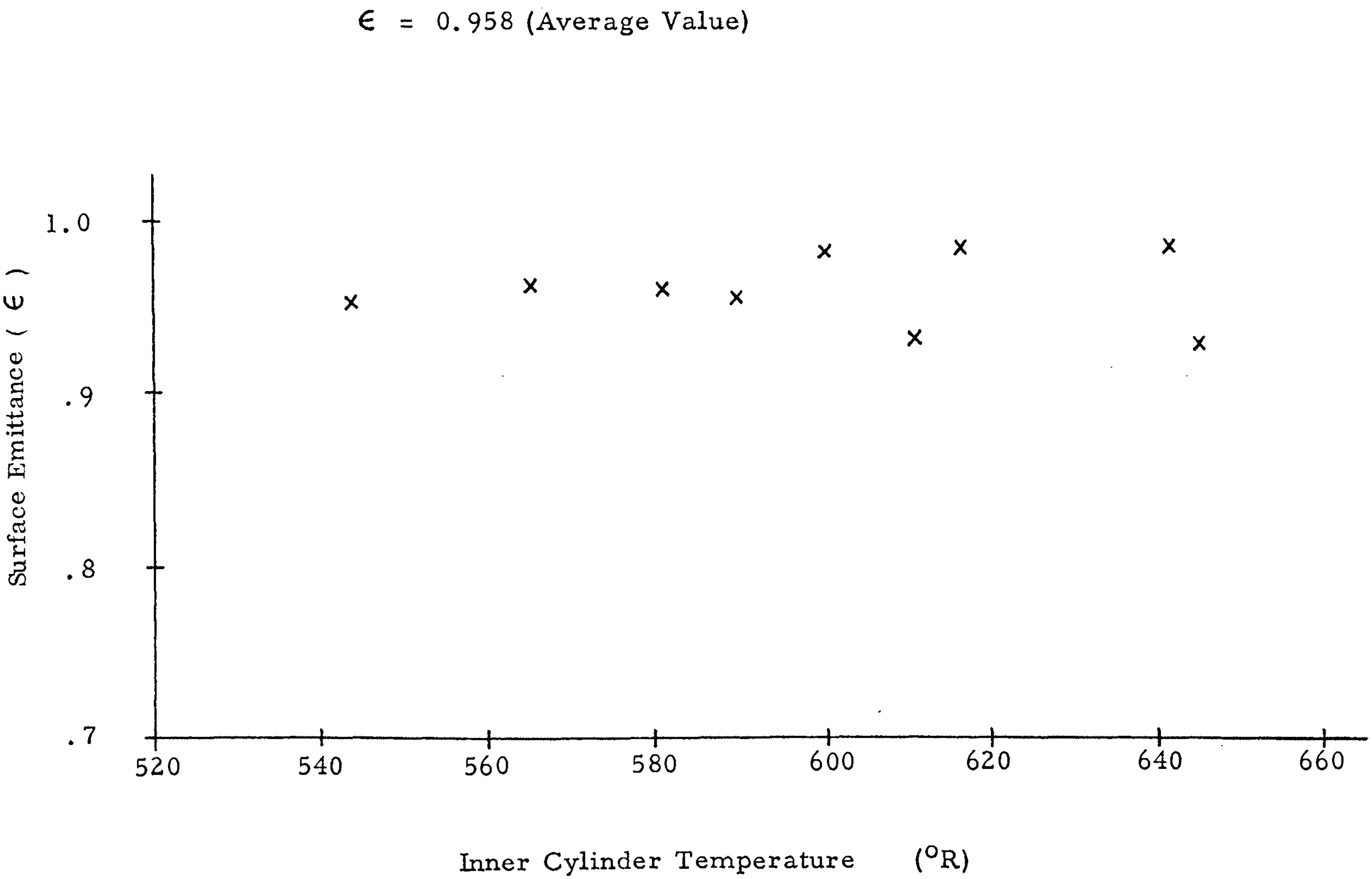


Fig. 5 Emittance of Black Painted Cylinder

α = 0.963 (Average Value)

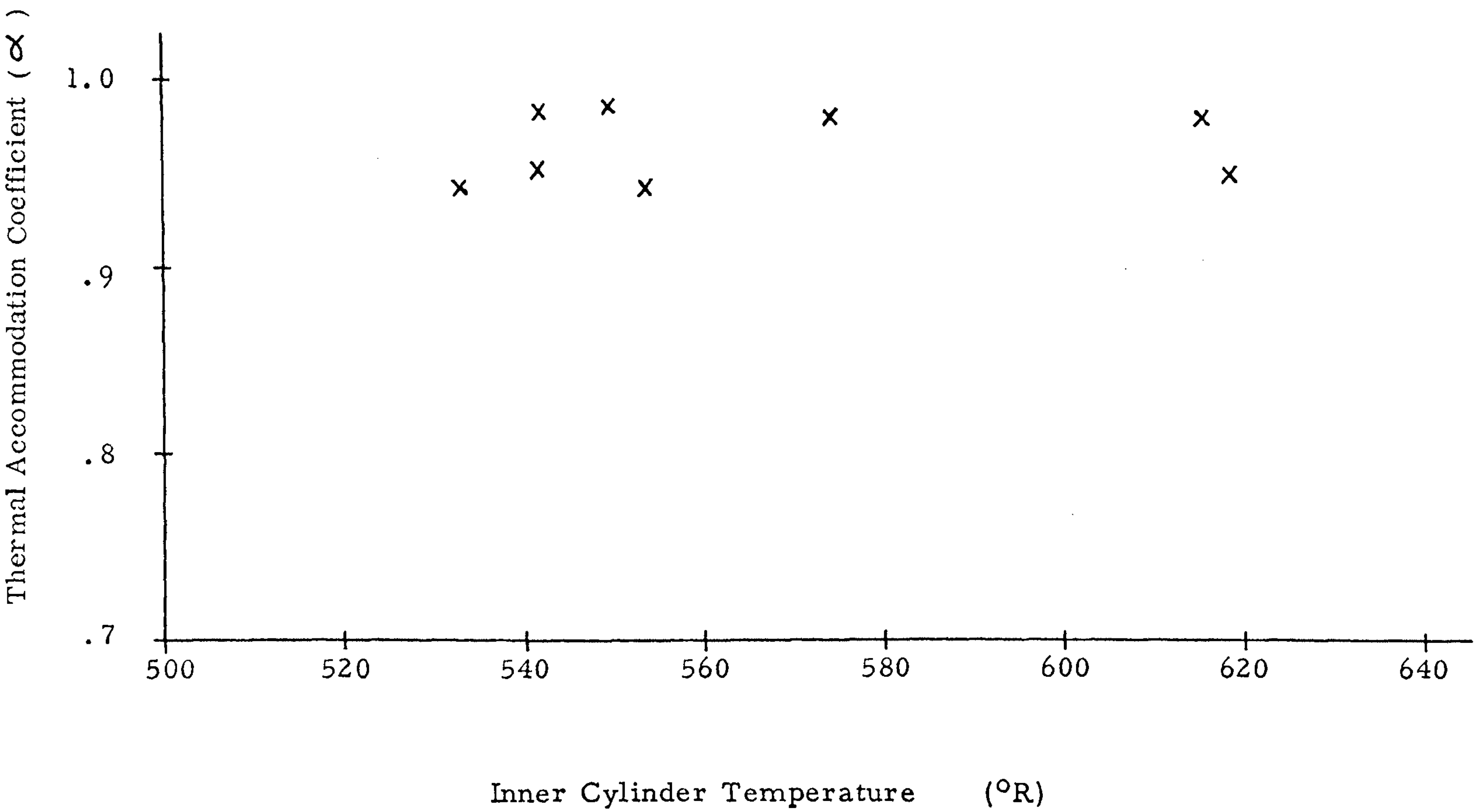


Fig. 6 Thermal Accommodation Coefficient of Black Painted Cylinder

thermal accommodation coefficient for the full range of temperature investigated, so the average value, $\alpha = 0.963$, was assumed constant.

B. ROUGH SURFACE MEASUREMENTS

Fig. 7 and Fig. 8 show the results from Table VI and Table VIII in Appendix A for the measured value of emittance for the steel cylinders with an average surface roughness of 25 and 7.5 microinches respectively. No meaningful change of emittance with temperature occurred for either test; however, emittance was effected by surface roughness as expected. For an average roughness of 25 microinches, $\epsilon_{avg.} = 0.174$, while for an average roughness of 7.5 microinches, $\epsilon_{avg.} = 0.123$. This decrease in emittance as the surface becomes more reflective is expected, and these values are typical for a steel of this composition (13).

Fig. 9 depicts the results from Table VII in Appendix A for the thermal accommodation coefficient for a surface with an average roughness of 25 microinches. The average value of the thermal accommodation coefficient for this condition is 0.835 with very little change due to temperature. Fig. 10 shows the results given in Table IX in Appendix A for the thermal accommodation coefficient where the average surface roughness is 7.5 microinches. The average value of the thermal accommodation coefficient for this condition is 0.693. Increased experimental error is seemingly present for this value.

The decrease in the thermal accommodation coefficient as the surface roughness is decreased can be accounted for by the surface interaction of

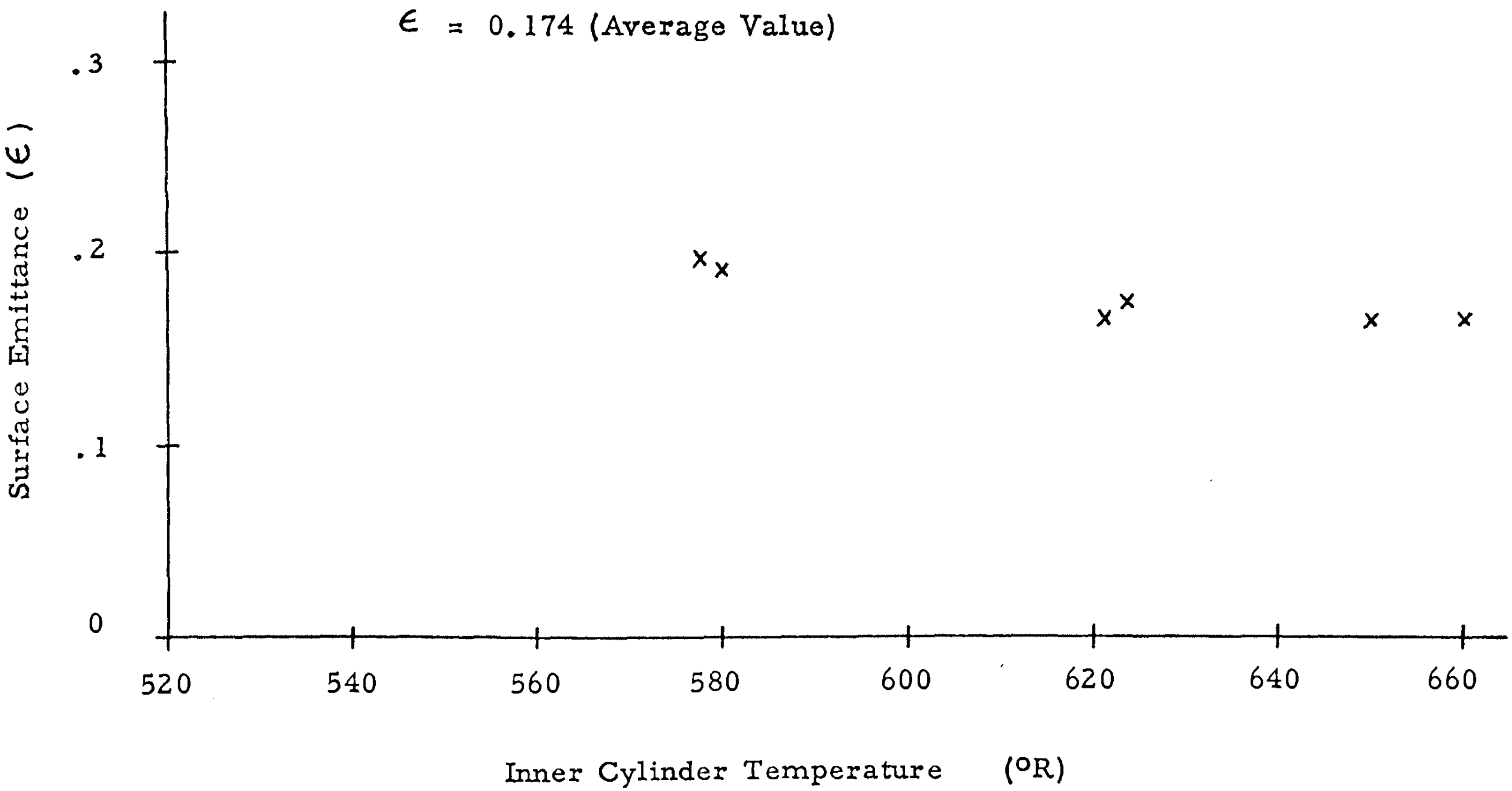


Fig. 7 Emittance of Steel Cylinder with Surface Roughness of 25 Microinches

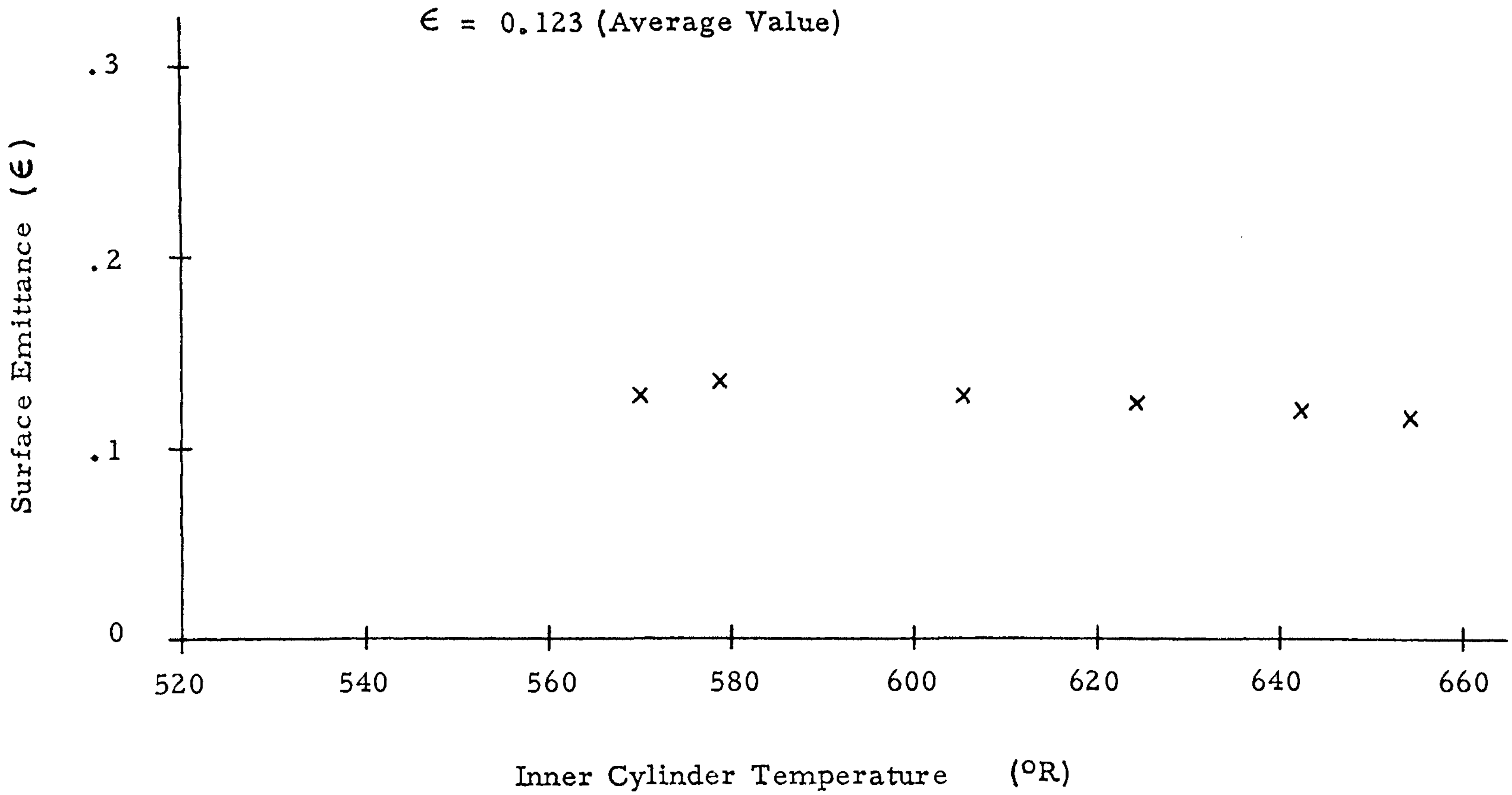


Fig. 8 Emittance of Steel Cylinder with Surface Roughness of 7.5 Microinches

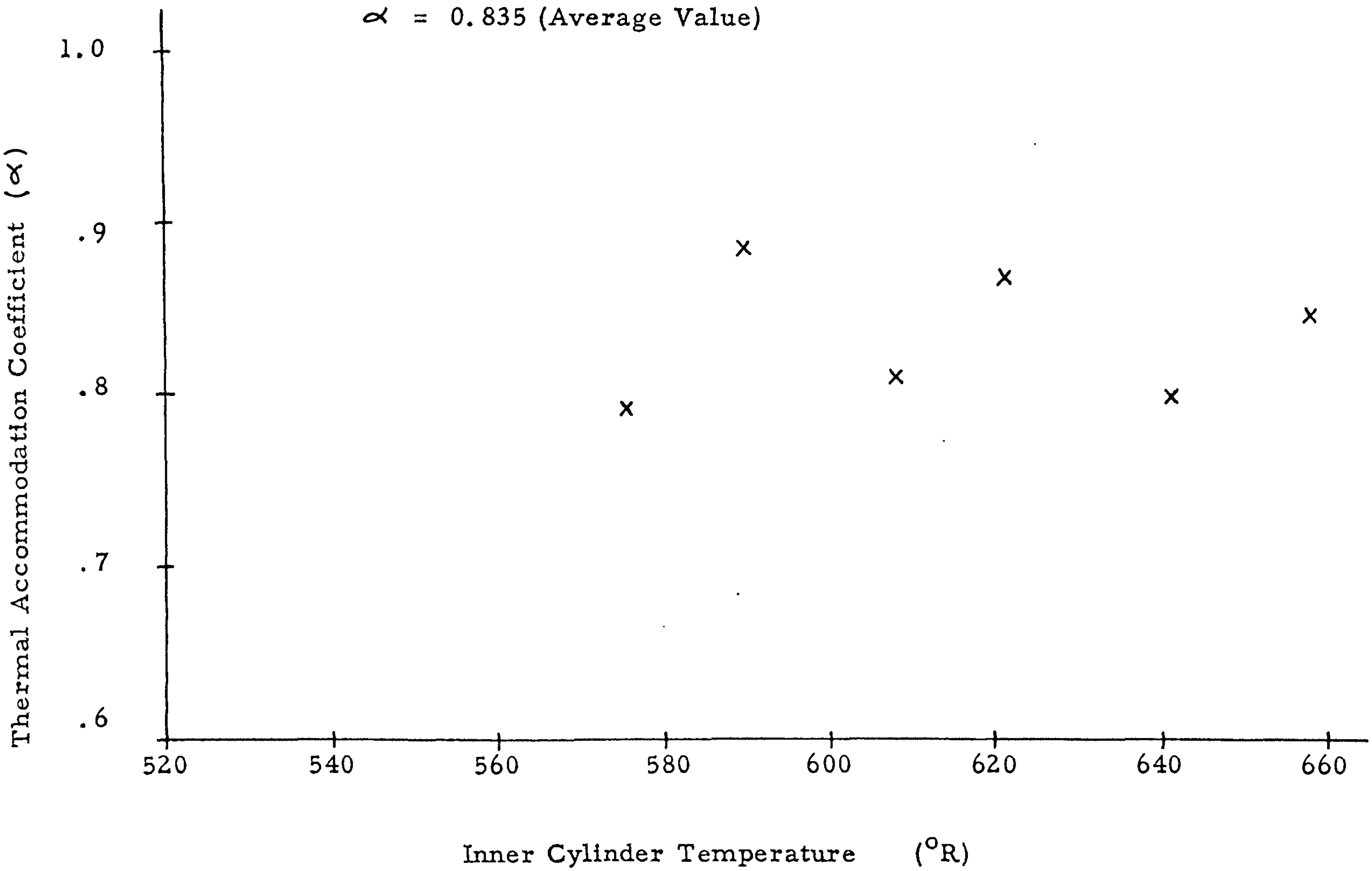


Fig. 9 Thermal Accommodation Coefficient of Steel Cylinder with Surface Roughness of 25 Microinches

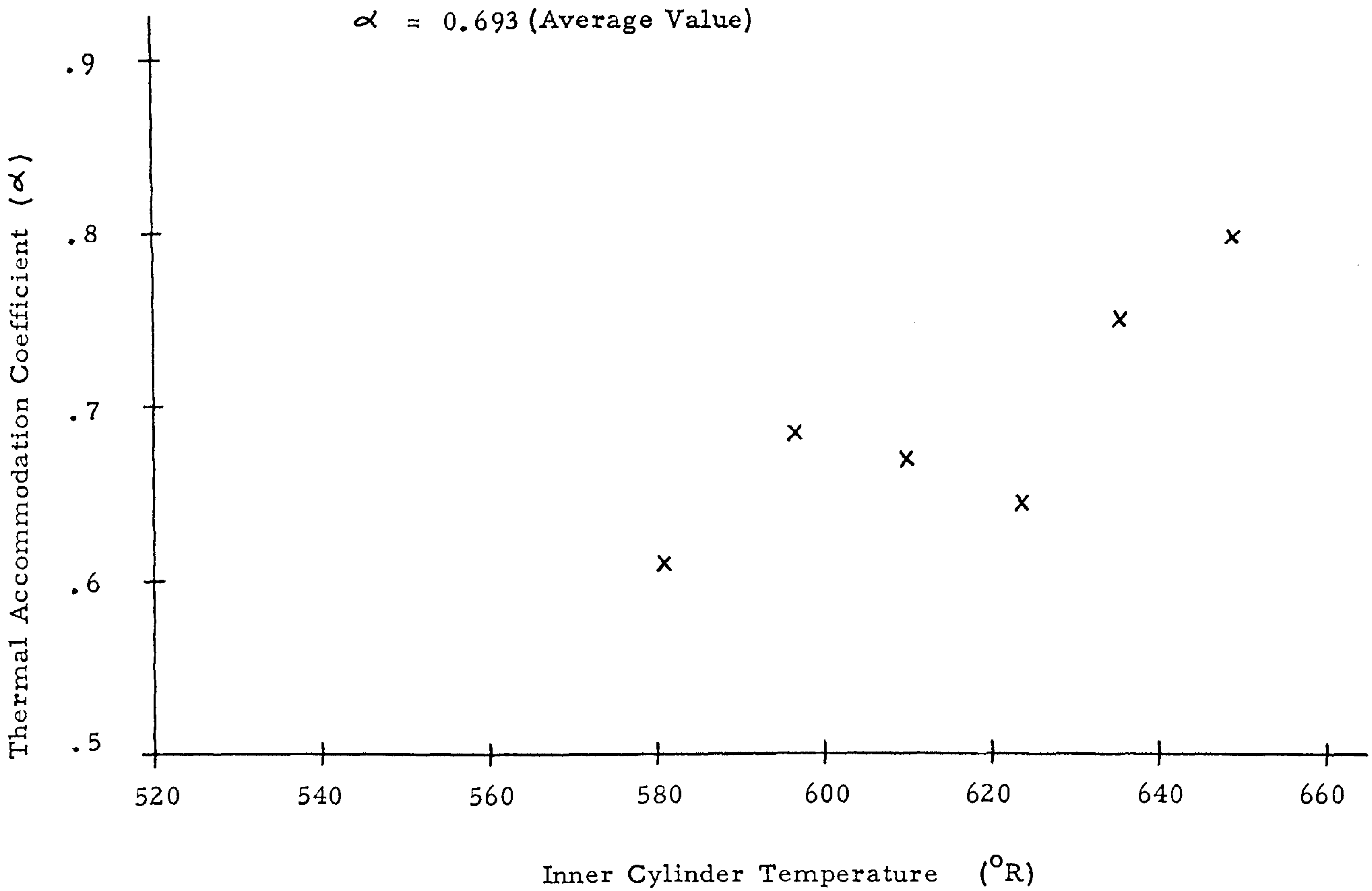


Fig. 10 Thermal Accommodation Coefficient of Steel Cylinder with Surface Roughness of 7.5 Microinches

a gas molecule while colliding. On the average, for a rougher surface, a molecule will make more than one collision with the surface molecules. The greater the number of collisions with the surface the greater the accommodation, thus a higher value of the thermal accommodation coefficient.

Table II compares the values of emittance and the thermal accommodation coefficient measured here with those obtained by Wiedmann and Trumpler (7) and Dethorne (2).

TABLE II
EXPERIMENTAL RESULTS COMPARED TO OTHER INVESTIGATIONS

		Wiedmann & Trumpler	Dethorne	Experiment
Black Surface	ϵ	0.932	0.965	0.958
	α	0.888	0.960	0.963
Steel Surface	ϵ	Not Conducted	0.1325 Surface Roughness (approximate 50 microinches)	0.1738 Surface Roughness (25 microinches) 0.1232 Surface Roughness (7.5 microinches)
	α	Not Conducted	0.971 Surface Roughness (approximate 50 microinches)	0.835 Surface Roughness (25 microinches) 0.693 Surface Roughness (7.5 microinches)

The maximum deviations of the experimental values from the average values are given in Table III.

TABLE III
MAXIMUM EXPERIMENTAL DEVIATIONS FROM AVERAGE VALUES

	Reference Surface		Steel Surface			
			Surface Roughness of 25 microinches		Surface Roughness of 7.5 microinches	
	Average Value	Maximum Deviation %	Average Value	Maximum Deviation %	Average Value	Maximum Deviation %
ϵ	0.958	3.5	0.174	12.6	0.123	7.5
α	0.963	2.4	0.835	10.6	0.693	15.2

C. ACCURACY OF RESULTS

An analysis of the accuracy of this experimental configuration was performed by Dethorne (2). An additional analysis was performed for the outgassing of the rubber hose connecting the McLeod Gage to the vacuum system and this was found to be negligible (see Appendix C). For a pressure of 1.2 micron Hg., the error in pressure measurement was found to be 0.0131 micron Hg.

The error introduced by end cylinder conduction due to temperature differences between the end of the cylinder and the end plug is evaluated in Appendix Cand was used in all of the present calculations.

Fig. 11 depicts the quantity $Q_C \sqrt{T_g} \;/\; (T_1 - T_2)$ versus

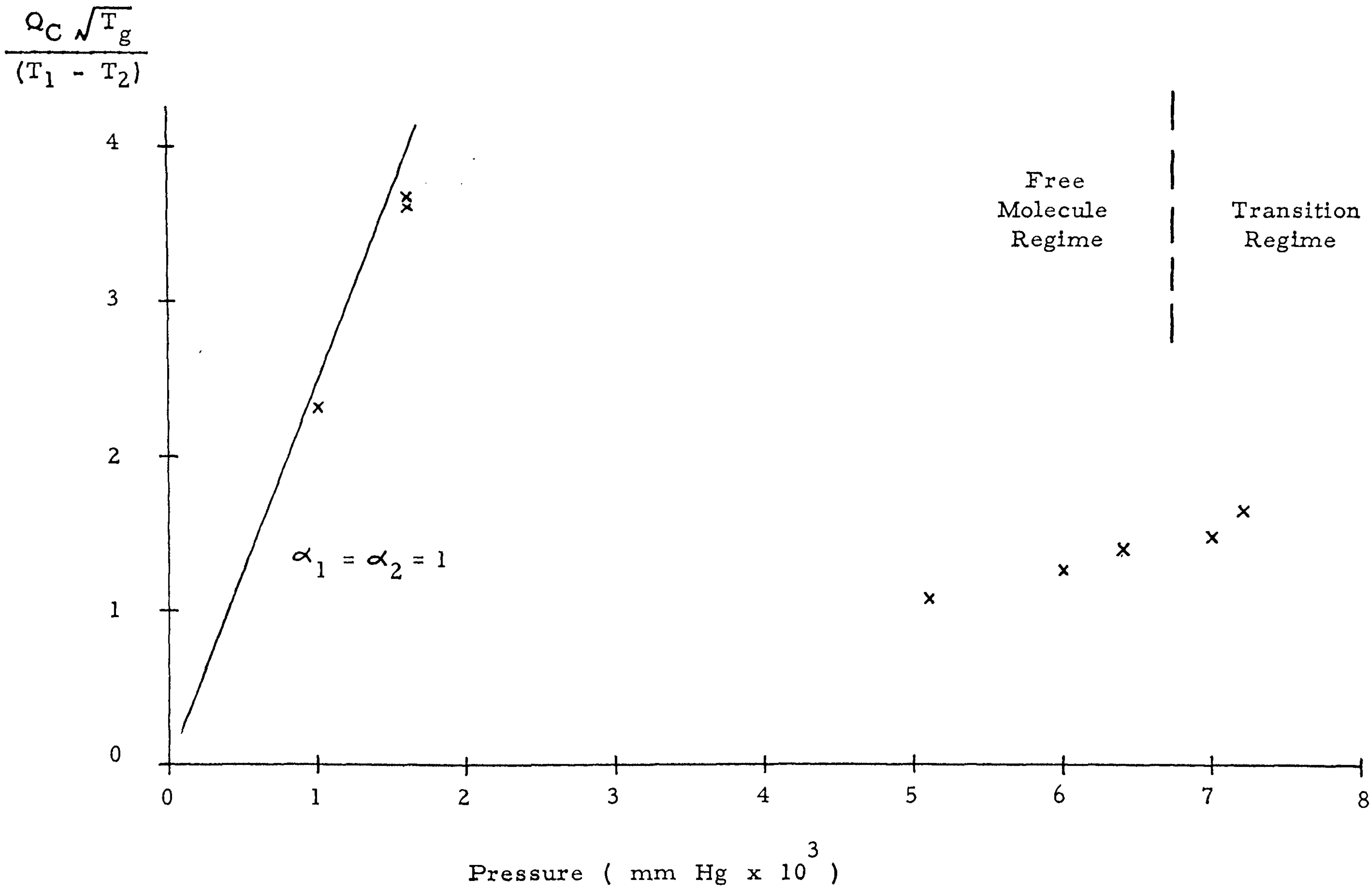


Fig. 11 Conduction Data for Black Painted Surface

pressure for the reference surface tests. This quantity is obtained from equation (21) and should be a linear function with the constant determined from the type of gas, emittance, and dimensions of the test unit. The line relating these parameters if $\alpha_1 = \alpha_2 = 1.0$ is also indicated in this figure. Several of the test results lie very close to this line while others lie on the borderline for the transition regime.

Fig. 12 depicts the same characteristics as Fig. 11 and the 25 micro-inches roughness of steel surface is included. In this case, the experimental data is well represented by equation (21) and Q_C is truly a linear function of the pressure. Similar results for the 7.5 microinches roughness of steel surface are shown in Fig. 13 with good agreement again being shown.

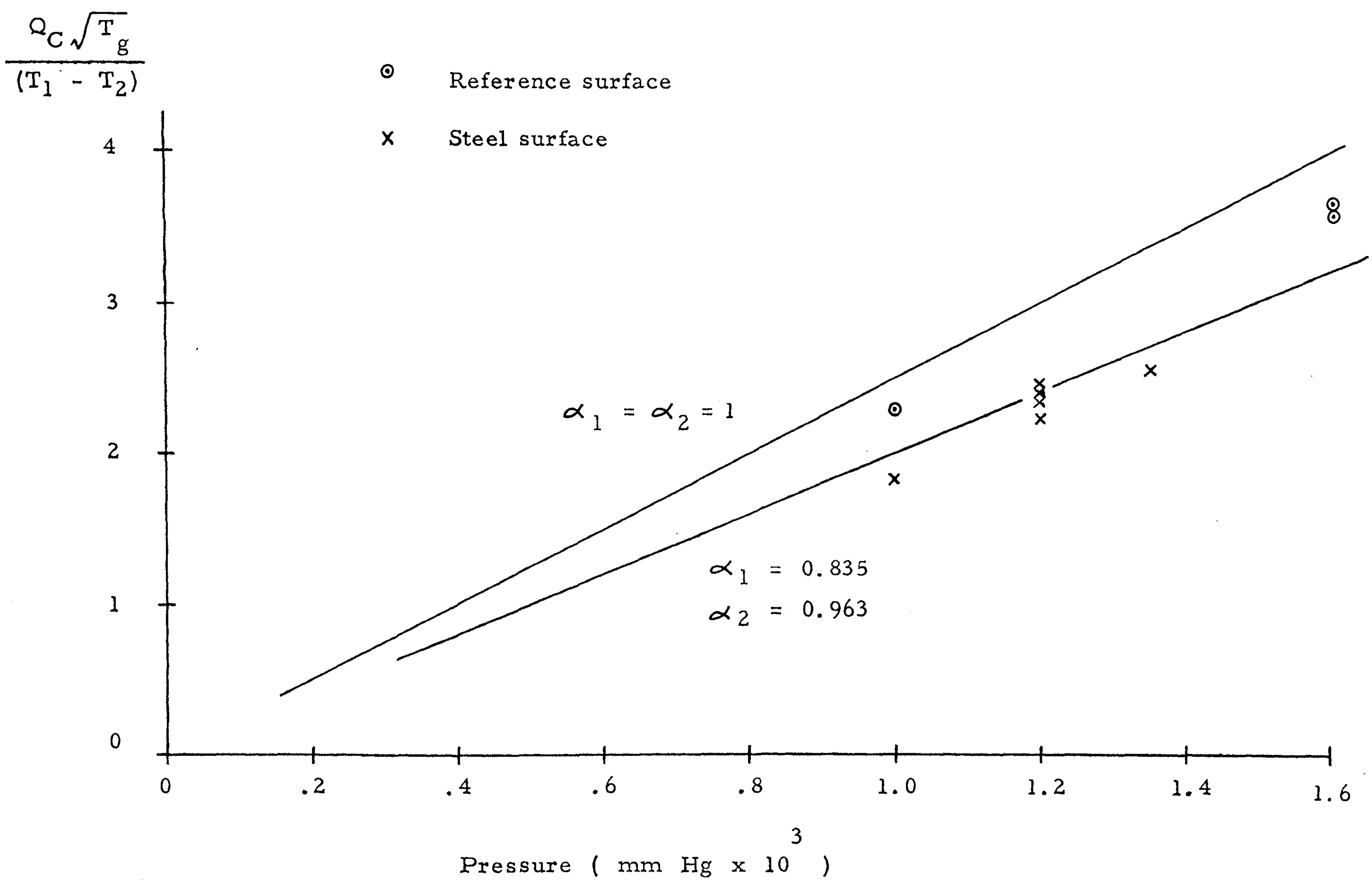


Fig. 12 Conduction Data for Steel Surface with Surface Roughness of 25 Microinches and Black Painted Surface

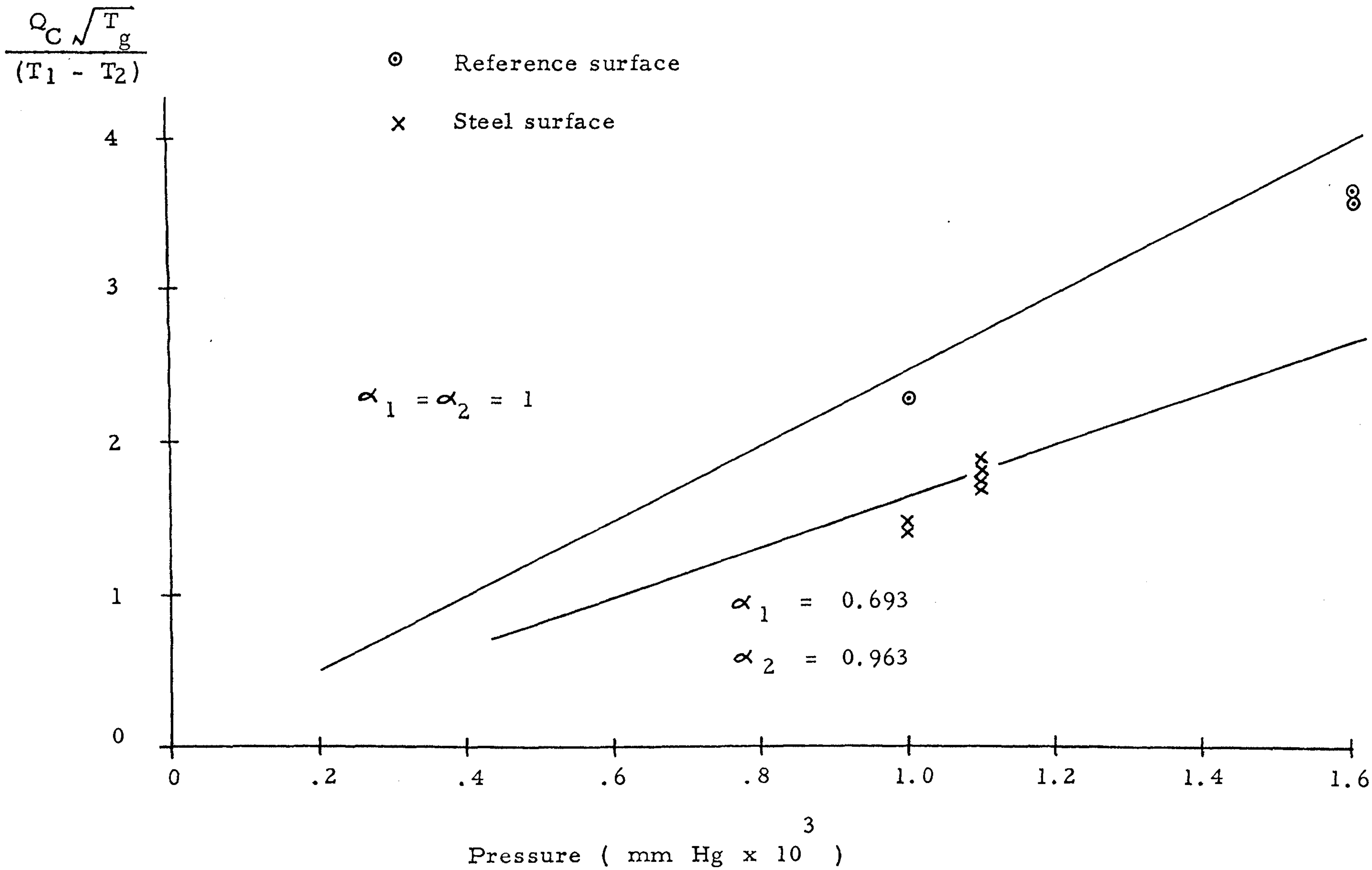


Fig. 13 Conduction Data of Steel Surface with Surface Roughness of 7.5 Microinches and Black Painted Surface

VI. CONCLUSIONS AND RECOMMENDATIONS

The following conclusions were arrived at from this investigation:

1. The value of the thermal accommodation coefficient and emittance given in Table II are reasonably accurate values.
2. These values of the thermal accommodation coefficient and emittance are in agreement with those presented by Wiedmann and Trumpler (7) and Dethorne (2) for regions where comparison is possible.
3. Increased surface roughness increases emittance as well as the thermal accommodation coefficient.
4. In the range of temperature from 75°F. to 200°F., there is no appreciable effect of temperature on the thermal accommodation coefficient.
5. Thermal conduction heat transfer is proportional to the pressure of the gas.

The following recommendations are made concerning future investigations:

1. A different apparatus design should be considered to reduce experimental time per test run and to improve the accuracy of the heat flow measurements. A device such as that used by Teagan and Springer (8) appears to be reasonable (guarded hot plate).
2. Different surface roughness values should be tested to obtain a relationship between the thermal accommodation coefficient and the average surface roughness.
3. Different engineering materials should be considered: aluminum,

copper, brass, painted surfaces.

4. Different gases should be used to find the effect of molecular weight on the thermal accommodation coefficient.

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VIII. VITA

The author, Wing On Ho, was born in Hong Kong on September 7, 1937. He received his high school education in Pui Ying Middle School in Hong Kong and graduated in July 1956. In the fall of the same year, he enrolled at Chung Chi College, Hong Kong, in the Physics Department. Upon graduation with a Bachelor of Science Degree in Physics in July 1960, he applied for admission to the University of Missouri, School of Mines and Metallurgy in the Mechanical Engineering Department. After graduation from the Missouri School of Mines and Metallurgy with a Bachelor Degree of Science in Mechanical Engineering 1964, he started working for Swindell-Dressler Company, a Division of Pullman Incorporated, in Pittsburgh, Pennsylvania as a Piping Engineer. In July 1966, he represented the company as Field Engineer at construction site for a Taconite Plant in Minnesota. Fifteen months later, he returned to the home office. In January 1968, he came to the University of Missouri - Rolla to continue his studies. From then until the present, he has been working toward the Degree of Master of Science in Mechanical Engineering.

The author is married to Hemeline M. M. Woo of Hong Kong and has one daughter Shanne.

APPENDIX A

EXPERIMENTAL DATA

The measurements of this experiment are tabulated in the following pages. The value of emittances determined from the radiation tests and the thermal accommodation coefficients determined from conduction tests are also included. The equations used for computations are shown in Appendix B.

TABLE IV

RADIATION TEST ON BLACK SURFACE

Outer Cylinder - Coated with Black Paint

Inner Cylinder - Coated with Black Paint (Average Surface Roughness = 40 microinches)

	RUN No. 2	RUN No. 3	RUN No. 5
Mean Free Path - INCH	2493.47700	2767.59900	2131.96200
Knudsen Number	68127.75000	73131.12000	58250.35000
Pressure - MM Mercury	0.000000755	0.0000007	0.00000088
Temperature - Inner Cylinder Deg. R	543.70000	580.60000	644.50000
Temperature - Outer Cylinder Deg. R	525.50000	523.00000	523.70000
Voltage - Volt	68.00000	133.00000	211.00000
Current - Ampere	0.02800	0.05070	0.07500
Total Heat Supply - BTU/HR	6.49816	23.01352	54.00912
Conductive Heat Flow - BTU/HR	0.00014	0.00042	0.00111
Radiative Heat Flow - BTU/HR	6.49801	23.01309	54.00801
Surface Emittance - Outer Cylinder	0.95103	0.095866	0.92517

TABLE IV (Continued)
RADIATION TEST ON BLACK SURFACE

Outer Cylinder - Coated with Black Paint
Inner Cylinder - Coated with Black Paint (Average Surface Roughness = 40 microinches)

	RUN No. 7	RUN No. 8	RUN No. 16
Mean Free Path - INCH	504.64080	506.14160	978.57270
Knudsen Number	13788.00000	13829.00000	26736.96000
Pressure - MM Mercury	0.0000037	0.0000037	0.0000019
Temperature - Inner Cylinder Deg. R	565.00000	589.30000	610.50000
Temperature - Outer Cylinder Deg. R	521.20000	522.75000	519.00000
Voltage - Volt	112.50000	152.00000	180.00000
Current - Ampere	0.04350	0.05200	0.06050
Total Heat Supply - BTU/HR	16.70187	26.97556	37.16647
Conductive Heat Flow - BTU/HR	0.00169	0.00257	0.00181
Radiative Heat Flow - BTU/HR	16.70018	26.97298	37.16466
Surface Emittance - Outer Cylinder	0.95969	0.95389	0.92981

TABLE IV (Continued)

RADIATION TEST ON BLACK SURFACE

Outer Cylinder - Coated with Black Paint

Inner Cylinder - Coated with Black Paint (Average Surface Roughness = 40 microinches)

	RUN No. 20	RUN No. 21	RUN No. 22
Mean Free Path - INCH	3243.04200	3216.48200	3237.17400
Knudsen Number	88607.68000	87882.00000	88447.37000
Pressure - MM Mercury	0.00000058	0.00000058	0.00000058
Temperature - Inner Cylinder Deg. R	616.00000	641.05000	600.20000
Temperature - Outer Cylinder Deg. R	525.05000	520.75000	524.10000
Voltage - Volt	182.00000	214.00000	160.00000
Current - Ampere	0.06800	0.08100	0.06150
Total Heat Supply - BTU/HR	42.23805	59.15915	33.58293
Conductive Heat Flow - BTU/HR	0.00055	0.00073	0.00046
Radiative Heat Flow - BTU/HR	42.23750	59.15842	33.58246
Surface Emittance - Outer Cylinder	0.98242	0.98183	0.97992

TABLE V

CONDUCTION TEST ON BLACK SURFACE

Outer Cylinder - Coated with Black Paint

Inner Cylinder - Coated with Black Paint (Average Surface Roughness = 40 microinches)

	RUN No. 10	RUN No. 11	RUN No. 12
Mean Free Path - INCH	1.87362	1.16765	1.17101
Knudsen Number	51.19177	31.90309	31.99483
Pressure - MM Mercury	0.00100	0.00160	0.00160
Temperature - Inner Cylinder Deg. R	541.80000	573.80000	615.10000
Temperature - Outer Cylinder Deg. R	523.00000	521.50000	523.00000
Voltage - Volt	75.00000	128.00000	189.00000
Current - Ampere	0.02700	0.04860	0.06500
Total Heat Supply - BTU/HR	6.91112	21.23096	41.92746
Radiative Heat Flow - BTU/HR	6.72249	20.39484	40.46371
Conductive Heat Flow - BTU/HR	0.18863	0.83612	1.46375
(Q * SQRT(T(2)) / (T(1) - T(2))	0.22946	0.36509	0.36346
Surface Emittance - Outer Cylinder	0.95807	0.95807	0.95807
Thermal Accommodation Coefficient - Outer Cylinder	0.98240	0.97955	0.97728

TABLE V (Continued)

CONDUCTION TEST ON BLACK SURFACE

Outer Cylinder - Coated with Black Paint

Inner Cylinder - Coated with Black Paint (Average Surface Roughness = 40 microinches)

	RUN No. 13	RUN No. 14	RUN No. 23
Mean Free Path - INCH	0.25873	0.26659	0.36790
Knudsen Number	7.06919	7.28375	10.05200
Pressure - MM Mercury	0.00720	0.00700	0.00510
Temperature - Inner Cylinder Deg. R	549.50000	618.00000	533.10000
Temperature - Outer Cylinder Deg. R	520.00000	520.90000	523.75000
Voltage - Volt	99.00000	200.00000	53.00000
Current - Ampere	0.03800	0.07200	0.02050
Total Heat Supply - BTU/HR	12.83933	49.14577	3.70811
Radiative Heat Flow - BTU/HR	10.69361	42.78101	3.26827
Conductive Heat Flow - BTU/HR	2.14572	6.36476	0.43984
(Q * SQRT(T(2)) / (T(1) - T(2))	1.65864	1.49603	1.07656
Surface Emittance - Outer Cylinder	0.95807	0.95807	0.95807
Thermal Accommodation Coefficient - Outer Cylinder	0.98440	0.94629	0.94002

TABLE V (Continued)
CONDUCTION TEST ON BLACK SURFACE

Outer Cylinder - Coated with Black Paint
Inner Cylinder - Coated with Black Paint (Average Surface Roughness = 40 microinches)

	RUN No. 24	RUN No. 25
Mean Free Path - INCH	0.29253	0.31227
Knudsen Number	7.99260	8.53196
Pressure - MM Mercury	0.00640	0.00600
Temperature - Inner Cylinder Deg. R	541.80000	553.50000
Temperature - Outer Cylinder Deg. R	522.60000	523.00000
Voltage - Volt	81.00000	100.00000
Current - Ampere	0.02900	0.03800
Total Heat Supply - BTU/HR	8.01690	12.96902
Radiative Heat Flow - BTU/HR	6.85782	11.27509
Conductive Heat Flow - BTU/HR	1.15908	1.69393
(Q * SQRT(T(2))/(T(1) - T(2))	1.38006	1.27012
Surface Emittance - Outer Cylinder	0.95807	0.95807
Thermal Accommodation Coefficient - Outer Cylinder	0.95082	0.94145

TABLE VI.

RADIATION TEST ON STEEL SURFACE WITH SURFACE ROUGHNESS OF 25 MICROINCHES

Outer Cylinder - Coated with Black Paint

Inner Cylinder - Machined Steel (Average Surface Roughness = 25 microinches)

	RUN No. 28	RUN No. 29	RUN No. 30
Mean Free Path - INCH	1039.50600	1039.90400	1036.91900
Knudsen Number	28132.79000	28143.56000	28062.76000
Pressure - MM Mercury	0.0000018	0.0000018	0.0000018
Temperature - Inner Cylinder Deg. R	577.90000	621.70000	660.20000
Temperature - Outer Cylinder Deg. R	522.30000	522.50000	521.00000
Voltage - Volt	59.00000	79.00000	97.00000
Current - Ampere	0.02300	0.02900	0.03600
Total Heat Supply - BTU/HR	4.63130	7.81895	11.91784
Conductive Heat Flow - BTU/HR	0.00104	0.00186	0.00261
Radiative Heat Flow - BTU/HR	4.63026	7.81709	11.91523
Surface Emittance - Outer Cylinder	0.95807	0.95807	0.95807
Surface Emittance - Inner Cylinder	0.19566	0.16357	0.16046

TABLE VI. (Continued)

RADIATION TEST ON STEEL SURFACE WITH SURFACE ROUGHNESS OF 25 MICROINCHES

Outer Cylinder - Coated with Black Paint

Inner Cylinder - Machined Steel (Average Surface Roughness = 25 microinches)

	RUN No. 31	RUN No. 32	RUN No. 33
Mean Free Path - INCH	1246.09400	1253.85500	1244.66100
Knudsen Number	33723.82000	33933.85000	33685.02000
Pressure - MM Mercury	0.0000015	0.0000015	0.0000015
Temperature - Inner Cylinder Deg. R	580.30000	624.10000	650.30000
Temperature - Outer Cylinder Deg. R	521.75000	525.00000	521.15000
Voltage - Volt	60.00000	82.00000	92.00000
Current - Ampere	0.02300	0.03000	0.03450
Total Heat Supply - BTU/HR	4.70980	8.39573	10.83254
Conductive Heat Flow - BTU/HR	0.00092	0.00155	0.00202
Radiative Heat Flow - BTU/HR	4.70888	8.39418	10.83052
Surface Emittance - Outer Cylinder	0.95807	0.95807	0.95807
Surface Emittance - Inner Cylinder	0.18790	0.17367	0.16144

TABLE VII.
CONDUCTION TEST ON STEEL SURFACE WITH SURFACE ROUGHNESS OF 25 MICROINCHES

Outer Cylinder - Coated with Black Paint
Inner Cylinder - Machined Steel (Average Surface Roughness = 25 microinches)

	RUN No. 34	RUN No. 35	RUN No. 36
Mean Free Path - INCH	1.86144	1.54791	1.39025
Knudsen Number	50.37724	41.89214	37.62526
Pressure - MM Mercury	0.00100	0.00120	0.00135
Temperature - Inner Cylinder Deg. R	575.50000	589.60000	607.60000
Temperature - Outer Cylinder Deg. R	519.60000	518.50000	523.90000
Voltage - Volt	59.00000	68.00000	79.00000
Current - Ampere	0.02250	0.02650	0.02850
Total Heat Supply - BTU/HR	4.53062	6.15005	7.68414
Radiative Heat Flow - BTU/HR	4.08135	5.38634	6.76024
Conductive Heat Flow - BTU/HR	0.44927	0.76371	0.92390
(Q * SQRT(T(2)) / (T(1) - T(2))	0.18320	0.24459	0.25265
Surface Emittance - Outer Cylinder	0.95807	0.95807	0.95807
Surface Emittance - Inner Cylinder	0.17378	0.17378	0.17378
Thermal Accommodation Coefficient - Outer Cylinder	0.96278	0.96278	0.96278
Thermal Accommodation Coefficient - Inner Cylinder	0.79437	0.88673	0.81200

TABLE VII. (Continued)
CONDUCTION TEST ON STEEL SURFACE WITH SURFACE ROUGHNESS OF 25 MICROINCHES

Outer Cylinder - Coated with Black Paint
Inner Cylinder - Machined Steel (Average Surface Roughness = 25 microinches)

	RUN No. 37	RUN No. 38	RUN No. 39
Mean Free Path - INCH	1.55030	1.55000	1.54791
Knudsen Number	41.95679	41.94867	41.89214
Pressure - MM Mercury	0.00120	0.00120	0.00120
Temperature - Inner Cylinder Deg. R	620.40000	640.90000	657.60000
Temperature - Outer Cylinder Deg. R	519.30000	519.20000	518.50000
Voltage - Volt	85.00000	95.00000	105.00000
Current - Ampere	0.03250	0.03650	0.03950
Total Heat Supply - BTU/HR	9.42813	11.83423	14.15500
Radiative Heat Flow - BTU/HR	8.36427	10.65202	12.72318
Conductive Heat Flow - BTU/HR	1.06386	1.18221	1.43182
(Q * SQRT(T(2)) / (T(1) - T(2))	0.23980	0.22135	0.23439
Surface Emittance - Outer Cylinder	0.95807	0.95807	0.95807
Surface Emittance - Inner Cylinder	0.17378	0.17378	0.17378
Thermal Accommodation Coefficient - Outer Cylinder	0.96278	0.96278	0.96278
Thermal Accommodation Coefficient - Inner Cylinder	0.86880	0.79996	0.84859

TABLE VIII.

RADIATION TEST ON STEEL SURFACE WITH SURFACE ROUGHNESS OF 7.5 MICROINCHES

Outer Cylinder - Coated with Black Paint

Inner Cylinder - Machined Steel (Average Surface Roughness = 7.5 microinches)

	RUN No. 40	RUN No. 41	RUN No. 42
Mean Free Path - INCH	1036.52100	1045.27800	1246.09400
Knudsen Number	27633.21000	27866.67000	33220.35000
Pressure - MM Mercury	0.0000018	0.0000018	0.0000015
Temperature - Inner Cylinder Deg. R	578.50000	605.60000	624.20000
Temperature - Outer Cylinder Deg. R	520.80000	525.20000	521.75000
Voltage - Volt	48.50000	60.00000	68.00000
Current - Ampere	0.02000	0.02300	0.02600
Total Heat Supply - BTU/HR	3.31051	4.70980	6.03401
Conductive Heat Flow - BTU/HR	0.00108	0.00151	0.00160
Radiative Heat Flow - BTU/HR	3.30943	4.70829	6.03240
Surface Emittance - Outer Cylinder	0.95807	0.95807	0.95807
Surface Emittance - Inner Cylinder	0.13480	0.12611	0.12146

TABLE VIII. (Continued)

RADIATION TEST ON STEEL SURFACE WITH SURFACE ROUGHNESS OF 7.5 MICROINCHES

Outer Cylinder - Coated with Black Paint

Inner Cylinder - Machined Steel (Average Surface Roughness =7.5 microinches)

	RUN No. 43	RUN No. 44	RUN NO. 45
Mean Free Path - INCH	1250.99000	1558.96100	1563.14100
Knudsen Number	33350.87000	41561.25000	41672.68000
Pressure - MM Mercury	0.0000015	0.0000012	0.0000012
Temperature - Inner Cylinder Deg. R	642.50000	654.60000	570.00000
Temperature - Outer Cylinder Deg. R	523.80000	522.20000	523.60000
Voltage - Volt	74.50000	80.00000	40.00000
Current - Ampere	0.02800	0.02900	0.01800
Total Heat Supply - BTU/HR	7.11931	7.91793	2.45729
Conductive Heat Flow - BTU/HR	0.00186	0.00166	0.00058
Radiative Heat Flow - BTU/HR	7.11745	7.91627	2.45671
Surface Emittance - Outer Cylinder	0.95807	0.95807	0.95807
Surface Emittance - Inner Cylinder	0.11703	0.11332	0.12647

TABLE IX.
CONDUCTION TEST ON STEEL SURFACE WITH SURFACE ROUGHNESS OF 7.5 MICROINCHES

Outer Cylinder - Coated with Black Paint

Inner Cylinder - Machined Steel (Average Surface Roughness =7.5 microinches)

	RUN No. 46	RUN No. 47	RUN No. 48
Mean Free Path - INCH	1.87075	1.69840	1.70557
Knudsen Number	49.87350	45.27873	45.46971
Pressure - MM Mercury	0.00100	0.00110	0.00110
Temperature - Inner Cylinder Deg. R	581.00000	596.40000	610.70000
Temperature - Outer Cylinder Deg. R	522.20000	521.50000	523.70000
Voltage - Volt	51.00000	60.00000	64.00000
Current - Ampere	0.02000	0.02300	0.02600
Total Heat Supply - BTU/HR	3.48116	4.70980	5.67906
Radiative Heat Flow - BTU/HR	3.11700	4.13809	5.02956
Conductive Heat Flow - BTU/HR	0.36415	0.57170	0.64951
(Q * SQRT(T(2)) / (T(1) - T(2))	0.14152	0.17431	0.17085
Surface Emittance - Outer Cylinder	0.95807	0.95807	0.95807
Surface Emittance - Inner Cylinder	0.12320	0.12320	0.12320
Thermal Accommodation Coefficient - Outer Cylinder	0.96278	0.96278	0.96278
Thermal Accommodation Coefficient - Inner Cylinder	0.60990	0.68476	0.67083

TABLE IX. (Continued)

CONDUCTION TEST ON STEEL SURFACE WITH SURFACE ROUGHNESS OF 7.5 MICROINCHES

Outer Cylinder - Coated with Black Paint

Inner Cylinder - Machined Steel (Average Surface Roughness = 7.5 microinches)

	RUN No. 49	RUN No. 50	RUN No. 51
Mean Free Path - INCH	1.87792	1.87040	1.87183
Knudsen Number	50.06450	49.86394	49.90213
Pressure - MM Mercury	0.00100	0.00100	0.00100
Temperature - Inner Cylinder Deg. R	623.90000	635.50000	649.00000
Temperature - Outer Cylinder Deg. R	524.20000	522.10000	522.50000
Voltage - Volt	72.00000	78.00000	83.50000
Current - Ampere	0.02700	0.02950	0.03200
Total Heat Supply - BTU/HR	6.63468	7.85308	9.11927
Radiative Heat Flow - BTU/HR	5.98496	6.99199	8.10060
Conductive Heat Flow - BTU/HR	0.64972	0.86109	1.01867
(Q * SQRT(T(2)) / (T(1) - T(2))	0.14920	0.17350	0.18407
Surface Emittance - Outer Cylinder	0.95807	0.95807	0.95807
Surface Emittance - Inner Cylinder	0.12320	0.12320	0.12320
Thermal Accommodation Coefficient - Outer Cylinder	0.96278	0.96278	0.96278
Thermal Accommodation Coefficient - Inner Cylinder	0.64379	0.75159	0.79872

APPENDIX B

SAMPLE CALCULATIONS

(A) Heat transfer by radiation between two concentric cylinders can be calculated by Eq. (16),

$$Q_R = \frac{2 \pi r_1 L \sigma}{\frac{r_1}{r_2} \left(\frac{1}{\epsilon_2} - 1 \right) + \frac{1}{\epsilon_1}} \left(T_1^4 - T_2^4 \right)$$

For similar surfaces:

$$\epsilon_1 = \epsilon_2 = \frac{Q_R}{\frac{2 \pi r_1 r_2 L \sigma}{r_1 + r_2} \left(T_1^4 - T_2^4 \right) + \frac{r_1}{r_1 + r_2} Q_R}$$

$$= \frac{Q_R}{3.2756 \times 10^{-10} \left(T_1^4 = T_2^4 \right) + 0.4904 \; Q_R}$$

For different surfaces:

$$\epsilon_1 = \frac{Q_R}{2 \pi r_1 L \sigma \left(T_1^4 - T_2^4 \right) - \frac{r_1}{r_2} \left(\frac{1}{\epsilon_2} - 1 \right) Q_R}$$

$$= \frac{Q_R}{6.43 \times 10^{-10} \; (T_1^{\;4} - T_2^{\;4}) - 0.963 \; (\frac{1}{\epsilon_2} - 1) \; Q_R}$$

(B) Heat transfer by conduction in free molecules regime between two concentric cylinders can be calculated by Eq. (23),

$$Q_C = 3600 \; r_1 \; L \; \frac{k+1}{k-1} \; \frac{p \, (T_1 - T_2)}{\sqrt{MT_g}} \; \frac{1}{\frac{r_1}{r_2} \, (\frac{1}{\alpha_2} - 1) + \frac{1}{\alpha_1}}$$

For similar surfaces:

$$\alpha_1 = \alpha_2 = \frac{(\frac{r_1}{r_2} + 1) \; Q_C}{3600 \; r_1 \; L \; \frac{k+1}{k-1} \; \frac{p \, (T_1 - T_2)}{\sqrt{MT_g}} + \frac{r_1}{r_2} \; Q_C}$$

$$= \frac{1.963 \; Q_C}{1.278 \times 10^3 \; \frac{p \; (T_1 - T_2)}{\sqrt{MT_g}} + 0.963 \; Q_C}$$

For different surfaces:

$$\alpha_1 = \frac{Q_C}{3600 \; r_1 \; L \; \frac{k+1}{k-1} \; \frac{p \; (T_1 - T_2)}{\sqrt{MT_g}} - \frac{r_1}{r_2} \; (\frac{1}{\alpha_2} - 1) \; Q_C}$$

$$= \frac{Q_C}{1.278 \times 10^3 \dfrac{p\ (T_1 - T_2)}{\sqrt{MT_g}} - 0.963\ \left(\dfrac{1}{\alpha_2} - 1\right)\ Q_C}$$

APPENDIX C

DATA CORRECTIONS

(A) The heat transfer from the center cylinder to the end plugs through aluminum and steel centering pins can be calculated by Fourier conduction equation:

$$Q' = KA_c \frac{\Delta T}{X}$$

where

Q' = Heat Transfer by Conduction, BTU/HR

K = Conductivity of Material, BTU/HR/°F/FT

A_c = Cross Sectional Area = 0.00213 FT^2

X = Distance of Heat Transfer = 0.00521 FT

ΔT = Temperature Difference, °F

TABLE X
HEAT TRANSFER AT STEEL END

Temperature Difference °F	Q' BTU/HR
0.05	0.5318
0.10	1.0636
0.20	2.1271
0.30	3.1907
0.40	4.2542
0.50	5.3178
0.60	6.3813
0.70	7.4449
0.80	8.5085
0.90	9.5720
1.00	10.6356

TABLE XI
HEAT TRANSFER AT ALUMINUM END

Temperature Difference °F	Q' BTU/HR
0.05	2.4237
0.10	4.8476
0.20	9.6947
0.30	14.5421
0.40	19.3895
0.50	24.2368
0.60	29.0842
0.70	33.9316
0.80	38.7789
0.90	43.6263
1.00	48.4737

(B) The rubber hose connected from the McLeod gage to the Varian vacuum unit evolves some amount of gases, especially when they are new. The vapor pressure of rubber and outgassing rates are given in Table XX (14).

TABLE XII
VAPOR PRESSURE AND OUTGASSING RATES OF RUBBER

	Vapor Pressure at 20°C (Torr)	Outgassing Rate after 3-hr Pumping *lusec/cm^2
Neoprene (hycar)	4×10^{-3}	$1 \times 10^{-3} - 3 \times 10^{-3}$
Silicone Rubber	2×10^{-4}	$2 \times 10^{-4} - 7 \times 10^{-3}$

$$* \; 1 \text{ lusec} = \frac{1 \text{ liter} \times 1 \text{ micron Hg.}}{1 \text{ second}}$$